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AN INVESTIGATION OF SOME OF THE
CHARACTERISTICS OF A JERK PUMP
INJECTION SYSTEM FOR
DIESEL ENGINES
—•••••
ROBERT LEE MOORE

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An Investigation of Some of the Characteristics
of a Jerk Pump Injection System for Diesel Engines

By

Robert Lee Moore, Jr.
B.S. (United States Naval Academy) 1930

Thesis

Submitted in partial satisfaction of the requirements for the degree of

MASTER OF SCIENCE

in

Mechanical Engineering

in the

Graduate Division

of the

University of California

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STATEMENT OF PROBLEM

The purpose of this investigation was to determine the influence of pump speed, nozzle opening pressure, rack setting, and length of line on the discharge rate of the injection valve in a jerk pump injection system for Diesel engines. Since some of the above variables may be controlled by an operator for any given system, it is felt that any correlation, mathematical or experimental, between these several variables would be of value to the designer of the injection system as well as to the Diesel operator.

In addition to obtaining a correlation between the above variables, FOR A PARTICULAR INSTALLATION, it appeared desirable to determine whether or not these variables might be changed with respect to one another to obtain performance which might be predicted, and which would result in better operation.

With these objectives in mind the problem was attacked from a purely experimental point of view and the results are recorded herein.

EXPERIMENTAL INVESTIGATION

The purpose of this investigation was to determine the influence of pump speed, water quality, and length of line on the efficiency of the injection pump. In a first series of tests (see Table I) the pump was operated at a constant speed of 1000 rpm. The water was of the same quality and the length of the line was varied. The results of these tests are shown in Table II. It is seen that the efficiency of the pump decreases as the length of the line increases. This is due to the fact that the friction loss in the line increases with the length of the line. The results of the tests are shown in Table III.

In addition to the above, a series of tests was conducted to determine the effect of water quality on the efficiency of the pump. The water was of two different qualities, one being of the same quality as the water used in the first series of tests, and the other being of a different quality. The results of these tests are shown in Table IV. It is seen that the efficiency of the pump is higher when the water is of the same quality as the water used in the first series of tests than when it is of a different quality. This is due to the fact that the friction loss in the line is lower when the water is of the same quality than when it is of a different quality.

With these results in mind, the pump was tested at a series of different speeds. The results of these tests are shown in Table V. It is seen that the efficiency of the pump increases as the speed increases. This is due to the fact that the friction loss in the line is lower at higher speeds than at lower speeds.

APPARATUS AND TEST PROCEDURE

A general view of the apparatus is shown in Figure 1, and a schematic diagram of the same apparatus is shown in Figure 4. In the above two figures the same letters are used to designate the same parts. Referring to either Figure 1 or Figure 4:

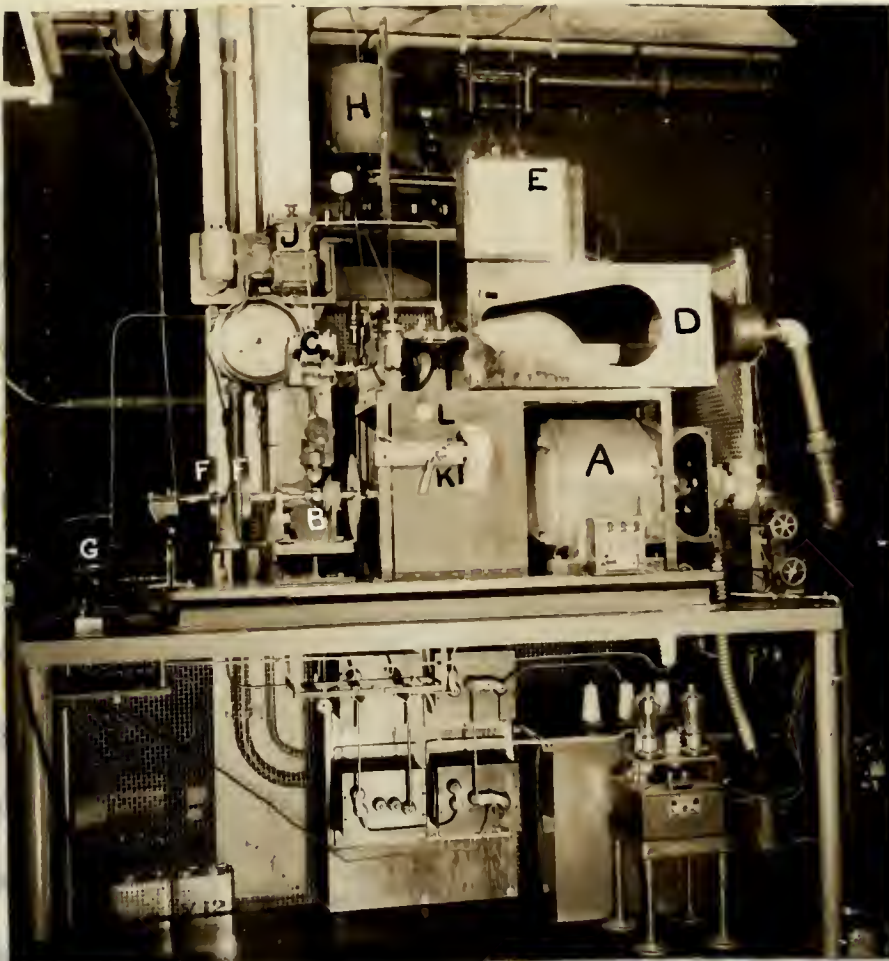


FIGURE 1.

"A" is a shunt motor having rheostatic speed control which drives the shaft to which is secured the cam "B", this cam in turn operates the plunger of the jerk pump "C". From the pump "C" the oil is discharged through a pipe line to the nozzle valve which is located in the box "D". A graduated scale is secured in box "D" as shown, by means of which the spray penetration may be determined. Mounted on

A general view of the structure is shown in Figure 1, and a
 detailed diagram of the same apparatus is shown in Figure 2. In
 the above two figures the main features are used to designate the
 same parts. Indicated in Figure 1 are Figure 2.



FIGURE 1.

"A" is a short metal having resistance equal to that of the
 the shell to which is secured the end "B", this end is then secured
 the bottom of the pipe joint "C". From the joint "C" the air is dis-
 charged through a pipe line to the inside valve which is located in
 the joint "D". A graduated scale is secured to the "D" as shown, by
 means of which the spray penetration may be determined. Indicated on

top of the box "D" is a stroboscopic neon light "E" which is made to illuminate the interior of the box "D" through a glass window in the top of "D". The time of the flashing of the neon light "E" may be made to occur at any desired angular shaft position by means of the graduated and variable rotary spark gap "F", one part of which is secured to the end of the motor shaft, while the variable part is mounted on the frame. A wiring diagram, Figure 5, shows the electrical connections between the spark gap "F" and the neon light "E". "H" is the oil reservoir located on one platform of a balance scale. Balance is obtained by placing weights of various magnitudes on the other platform; the instant of perfect balance being indicated by the flashing of a small neon light "L", Figure 1. This neon light is controlled by an electrical circuit through two small wires secured to either platform; the wires moving in and out of mercury baths, located under the platforms, as the balance changes position. From "H" the oil is delivered to the fuel pump "C" through the oil filter "J". It is to be noted that oil may be delivered to the pump "C" by gravity or under pressure by means of the variable speed pump "G". A pressure of 12 pounds per square inch was maintained at the suction side of the pump "C" throughout the investigation.

The pump used was a 10-millimeter "jerk-pump", having the conventional plunger scroll control. A cut-away picture of this type of pump is shown in Figure 2(B). The quantity of oil delivered by the pump may be varied by rotating the plunger "P" by means of the gear "G" which is secured to the plunger "P". Gear "G" is in turn

top of the box "1" is a cylindrical metal light "2" which is made
 to illuminate the interior of the box "1" through a glass window
 in the top of "1". The base of the cylinder of the light "2" is
 only for the purpose of the light being reflected in the
 of the cylinder and window which is only "3", the part of
 which is located in the box "1" and in the lower part, which is the window
 part is located in the box "1" and in the lower part, which is the window
 the electrical connections between the light "2" and the box
 light "2". "2" is the oil reservoir located at the bottom of a
 balance scale. Balance is obtained by placing weights on various
 weights on the other platform. The balance of the balance
 being indicated by the flashing of a small lamp light "4", figure 1.
 This lamp light is controlled by an electrical circuit through two
 small wires secured to either platform. The wires extend to and
 are at various points, located under the platform, at the bottom
 through platform. From "5" the oil is delivered to the fuel pump
 "6" through an oil filter "7". It is to be noted that all oil is
 delivered to the pump "6" by gravity or under pressure by means of
 the electric pump "8". A pressure of 15 pounds per square
 inch was maintained at the various ends of the pump "6" throughout
 the investigation.

The fuel used was a 10-40-40 motor "9", which the com-
 position of the fuel control. A fuel-air mixture of 10:1
 of fuel is shown in figure 1(a). The quantity of oil delivered by
 the pump was for testing in testing the pump "6" by means of the
 pump "6" which is shown in the diagram "7". Gas "9" is in test

actuated by means of an engaging rack, the position of which is controlled by the micrometer head "R", Figure 4. The end of the plunger "P" is actuated by the cam "B", Figure 4, by means of suitable linkage. The operation of the fuel pump "B", Figure 2, is as follows: as the plunger "P" moves from right to left, the space "E" becomes isolated when the flat plunger face reaches the left edge of the inlet channel "J". Further motion of "P" to the left causes the plunger to force oil from "E" past the check valve "V" until the left edge of the scroll space "S" reaches the right edge of the inlet channel "J". At this point the space "E" communicates directly with the inlet channel "J" through a small hole drilled from the plunger face to the scroll space "S", and the oil in space "E" is by-passed back to "J".

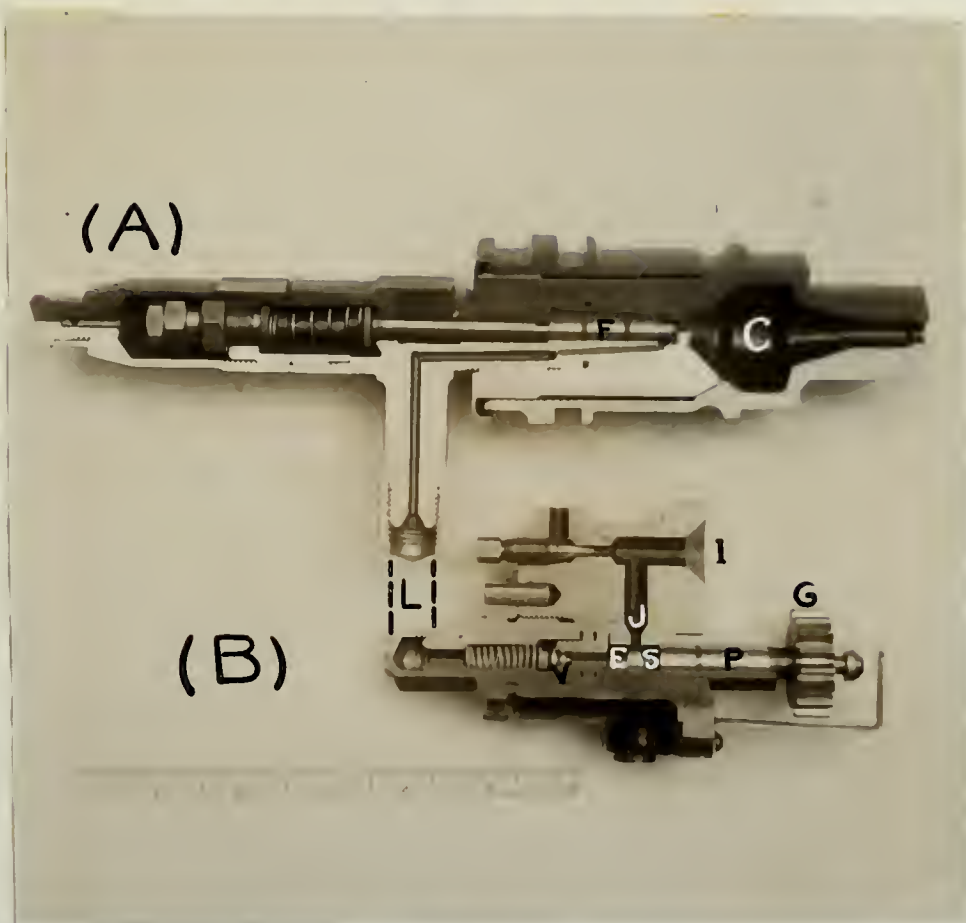


Figure 2.

entrance to each of the two pipes, the position of which is also
 marked by the dimension lines $2''$, figure 4. The end of the pipes
 $2''$ is indicated by the line $2''$, figure 4, by means of vertical lines.
 The operation of the two pipes $2''$, figure 4, is as follows: as the
 plunger $2''$ moves from right to left, the space $2''$ between the
 ends of the two pipes (the space between the left edge of the left channel
 $2''$, figure 4, and the left edge of the right channel $2''$, figure 4) is
 all then $2''$ past the point where $2''$ enters the left edge of the
 second space $2''$ reaches the right edge of the left channel $2''$. In
 this point the space $2''$ communicates directly with the left channel
 $2''$ through a small hole drilled from the plunger into the second
 space $2''$, and the air in space $2''$ is by-passed from $2''$.



Figure 2.

Thus in this type of pump, delivery will always commence at the same angular shaft position, regardless of rack setting, whereas the angular shaft position at which oil ceases to be delivered will depend upon the setting of the rack.

Oil enters the pump Figure 2(B) at "I", is discharged past the check valve "V" from whence it is delivered to the nozzle valve, Figure 2(A) through the pipe line "L".

A Caterpillar Fuel Injection Valve designed for engines of $5\frac{1}{4}$ " and $5\frac{3}{4}$ " cylinder bore was used in this investigation. A picture of this valve, in cut-away section, is shown in Figure 2(A); the valve actually used, however, differs from the one shown in Figure 2(A) in that the pre-combustion chamber "C" was not used. The manufacturer's part designation for this valve is as listed below:

<u>Part Name</u>	<u>Part Number</u>	<u>Weight (Grams)</u>
Spray valve spring	1A6926	11.22
Spray valve spring stem	2A4684	16.07
Spray valve needle	2A4682	5.74

Other pertinent data pertaining to the nozzle valve are:

- (1) needle valve lift 0.007"; (2) needle valve stem diameter 0.039";
- (3) included angle between faces of valve seat = 60° ; (4) orifice length 0.118"; (5) orifice diameter 0.025"; (6) spring constant 771.2 lbs. per in. deflection; (7) clearances between "P" and "F" and their respective working surfaces = lap fit.

In obtaining the data necessary for plotting the curves shown

in Figure 1, 2 and 3, the adjustable spring of the injection valve was fixed at 100 lb. and the air in a small pump to force the water to the valve of the injection pressure. The pump and the valve were at a constant value and the number of pump strokes per minute for the delivery of a known weight of oil, as indicated by the balance and injection light system, was recorded on the provided number "1". Figures 1 and 2. Several "runs" were made for each setting in order to obtain a series of data.

The spray operation shown in Figure 3 was taken at a pump speed of 800 R.P.M. and a tank setting of 100 (injection valve). In Figure 3 it was taken at the same pump speed but for a tank setting of 100 (light tank). The same operation pressure in both cases was 100 lb. per sq. in. Figures were taken for every one degree regular about displacement, starting with the first tank of 10-20 and continuing until the point of cut-off. Since a speed of 800 R.P.M. is equivalent to one revolution in a tank of a second, the photos were taken at time exposure of one-half of a second. The spray giving only one injection per picture rather than a series of injections. The series of pictures represent several hundred injections rather than a time development of one injection.

The characteristic properties of the diesel fuel used were as follows:

Gravity.....	50.8 A.P.I. at 60°F.
Viscosity.....	28.7 cP at 100°F.
Specific Gravity.....	0.868 at 60°F.

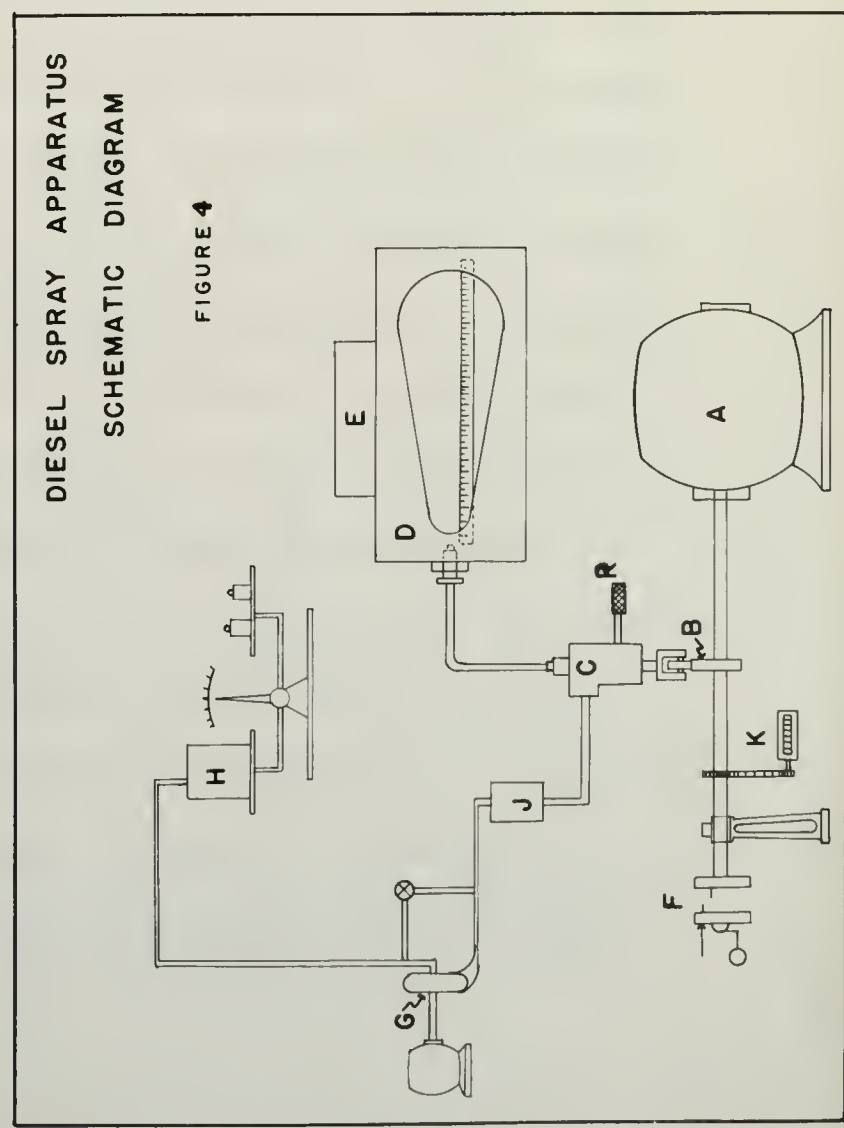
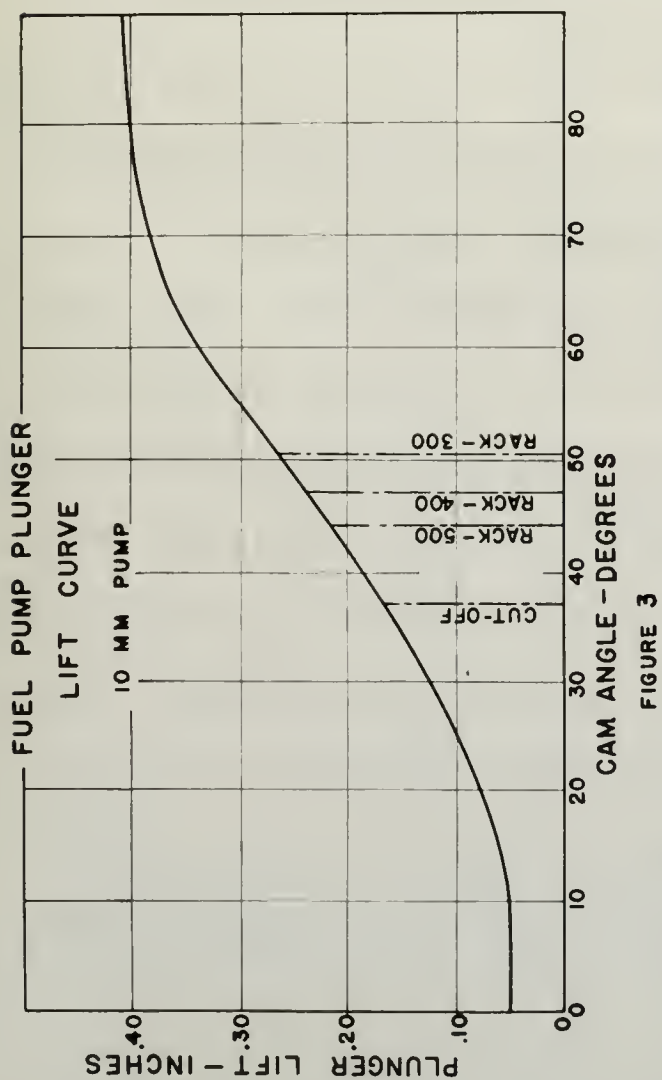


FIGURE 4

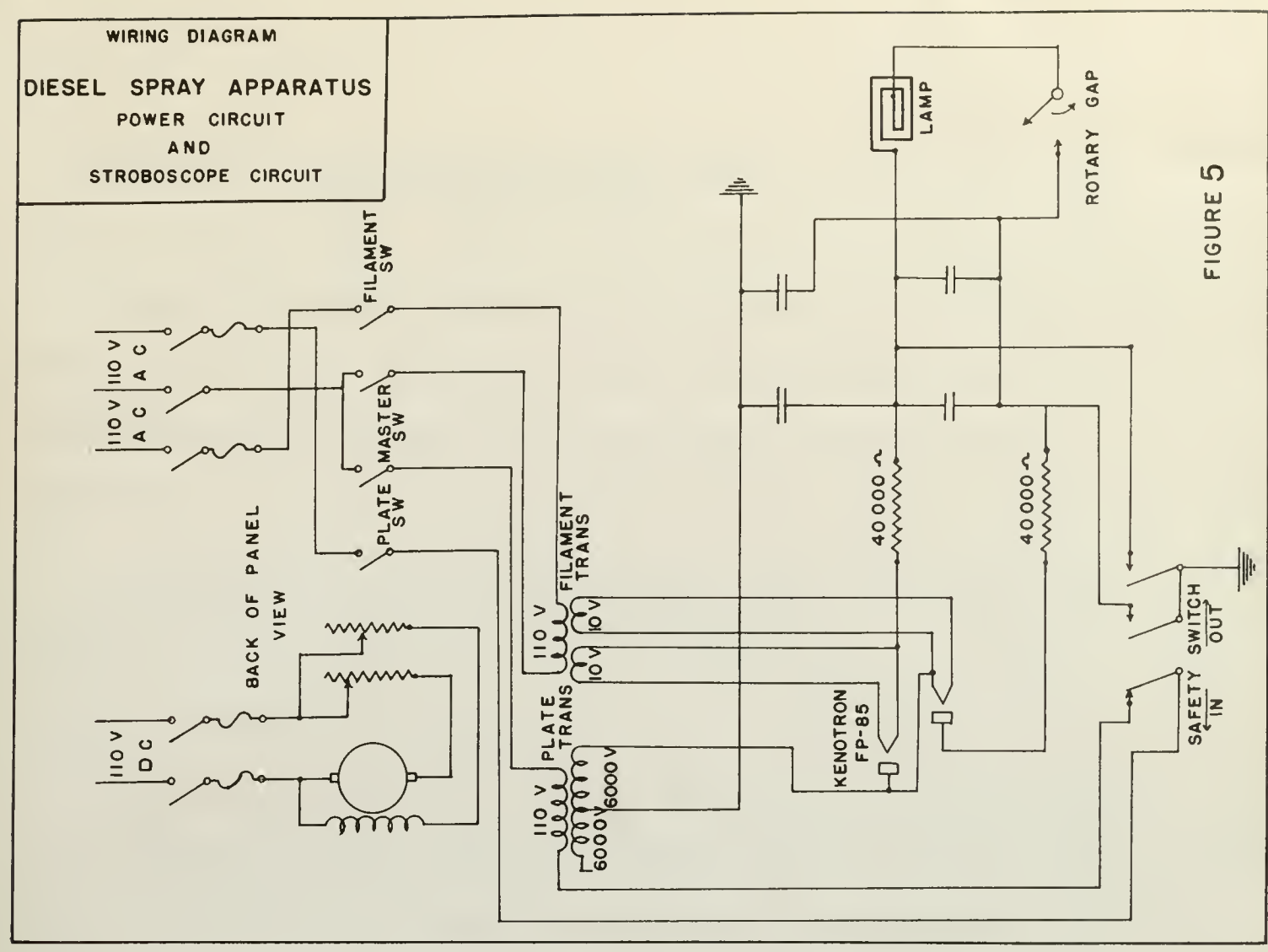


FIGURE 5

DISCUSSION AND RESULTS

The quantity of oil which the fuel pump, Figure 2(B), should deliver for various rack settings, based on displacement data, was computed as follows and is shown by the plane surfaces A B C D in Figure 6:

Dial indicator readings were carefully taken of the cam contour for the various angular crank positions from which the lift curve, Figure 3, was constructed. Then the angular shaft position for cut-off (the point at which the fuel pump, Figure 2(B), began delivery) was obtained by manually unseating the check valve and observing the point at which the oil ceased to flow from the fuel pump; the oil to the suction side of the pump being maintained under pressure due to the gravity head from supply tank "H", see Figure 1. This angular shaft position was, as should be expected, the same for all rack settings. Next, the angular position of the shaft at the point of release (that point at which the pump stopped delivery due to the scroll position), was determined for each rack position. This was accomplished by again lifting the fuel pump check valve manually, and observing the angular shaft position at which oil began to flow from the pump. These angular positions were then transferred to the lift curve, Figure 3, from which the effective pump stroke for any rack setting was obtained by subtracting the lift at the point of cut-off from the lift at the point of release. Knowing the effective pump stroke, for any rack setting, the pump plunger area and the density

The quantity of oil which the pump pump (Figure 1), would deliver for various work positions, based on displacement data, was computed as follows and is shown in the table between 100 and 150 cc.

Figure 2.

Dial indicator readings were carefully taken of the cam surface for the various angular work positions from which the lift came, Figure 3, was constructed. From the angular work position the lift-off (the point at which the lift pump, Figure 4), began delivery) was obtained by mentally measuring the angles which are shown in the point at which the oil began to flow from the lift pump. The oil for the suction side of the pump being contained under pressure due to the gravity head from supply tank "B", see Figure 1. This angular work position was, as should be expected, the same for all work positions. Next, the angular position of the shaft at the point of lift-off (that point at which the pump stopped delivery due to the supply position), was determined for each work position. This was determined by again lifting the lift pump check valve manually, and observing the angular shaft position at which oil began to flow from the pump. These angular positions were then transferred to the lift curve, Figure 3, from which the effective pump stroke for any work position was obtained by subtracting the lift at the point of cut-off from the lift at the point of release. Knowing the effective pump stroke, for any work position, the pump chamber area and the density

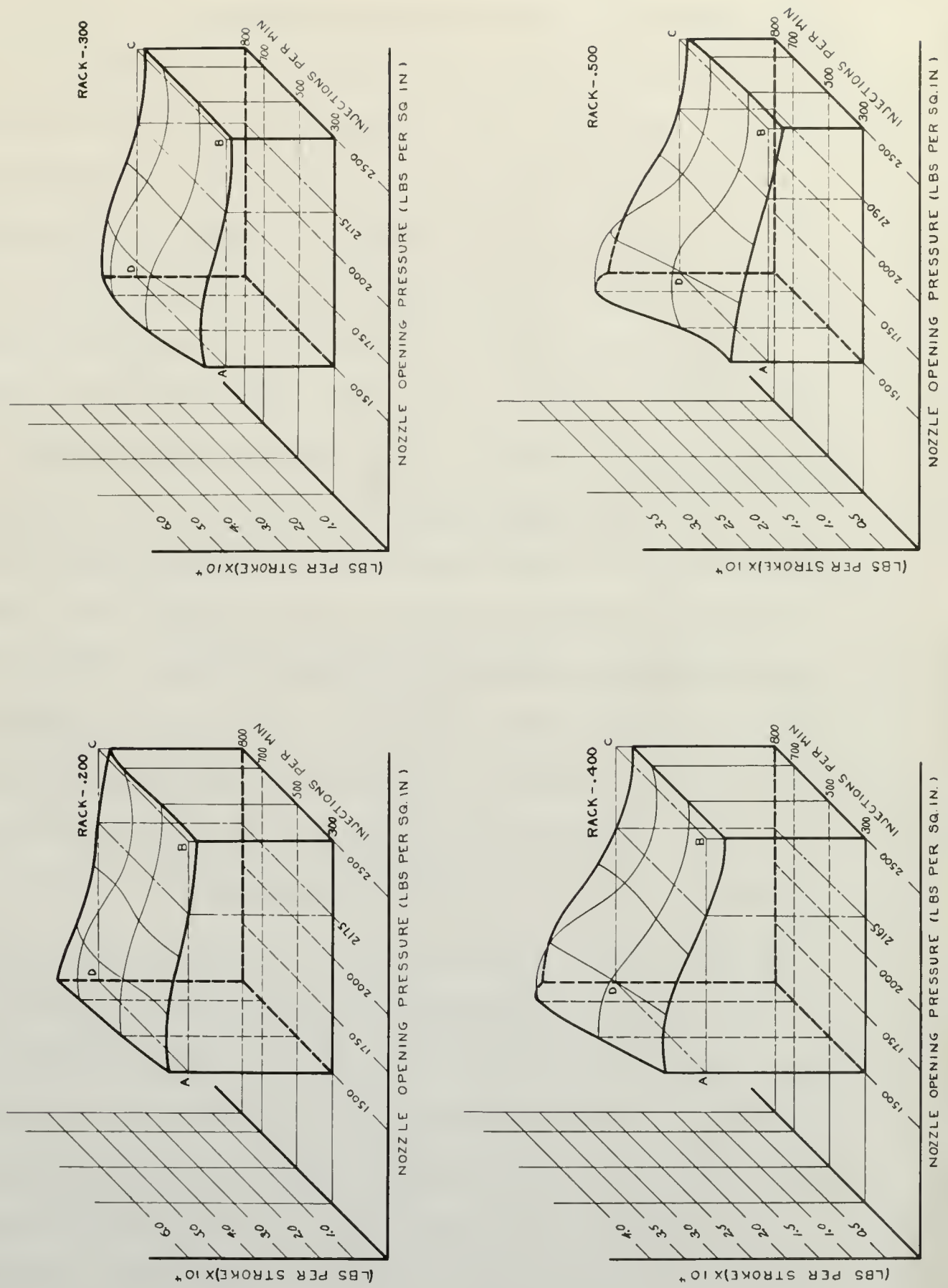
of the Diesel oil, the weight of oil discharged per stroke was computed from the formula: weight per stroke = area of piston x stroke of piston x oil density. Since the area of the pump was .000846 square feet (plunger diameter 10 millimeters), and the density of the oil at 12 lbs. per sq. in. pressure and 74°F, was 62.284 x .8676 = 54.038 pounds per cubic foot, the weight of oil delivered per stroke was: .000846 x 54.038 x $\frac{\text{lift}''}{12}$ = .00381 x lift'' (lbs.). The results of the above computations are as tabulated:

TABLE I

<u>Rack Setting</u>	<u>Cut Off</u>	<u>Re-lease</u>	<u>Lift at Cut Off</u>	<u>Lift at Release</u>	<u>Effective Pump Stroke</u>	<u>Calculated Wt. of oil per Stroke (lbs.)</u>
.150	376	56°	.175"	.305"	.130"	4.96 x 10 ⁻⁴
.175	"	55°	"	.297"	.122"	4.65 x 10 ⁻⁴
.200	"	54°	"	.290"	.115"	4.38 x 10 ⁻⁴
.225	"	53°	"	.282"	.107"	4.08 x 10 ⁻⁴
.250	"	52°	"	.274"	.099"	3.77 x 10 ⁻⁴
.300	"	50.5°	"	.262"	.087"	3.315 x 10 ⁻⁴
.350	"	49.25°	"	.253"	.078"	2.97 x 10 ⁻⁴
.400	"	47.25°	"	.238"	.063"	2.40 x 10 ⁻⁴
.450	"	46°	"	.227"	.052"	1.98 x 10 ⁻⁴
.500	"	44°	"	.213"	.038"	1.45 x 10 ⁻⁴
.550	"	42.5°	"	.202"	.027"	1.03 x 10 ⁻⁴
.600	"	40°	"	.185"	.010"	0.381 x 10 ⁻⁴

From Figure 6, it may be observed that the surface representing the oil actually discharged per stroke for any given rack setting

THE INFLUENCE OF PUMP SPEED AND NOZZLE
OPENING PRESSURE
ON DISCHARGE RATE FOR VARIOUS RACK SETTINGS
PIPE LENGTH - 30 INCHES
FIGURE 6

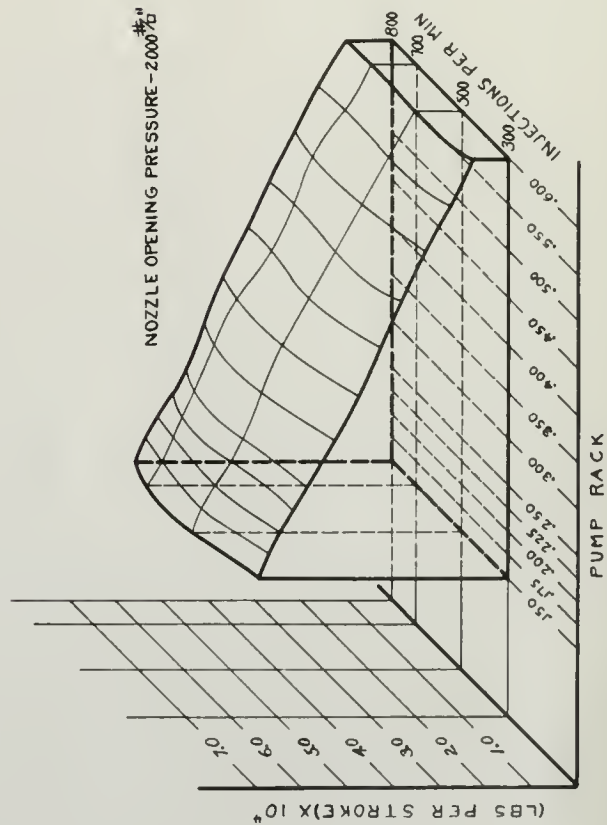
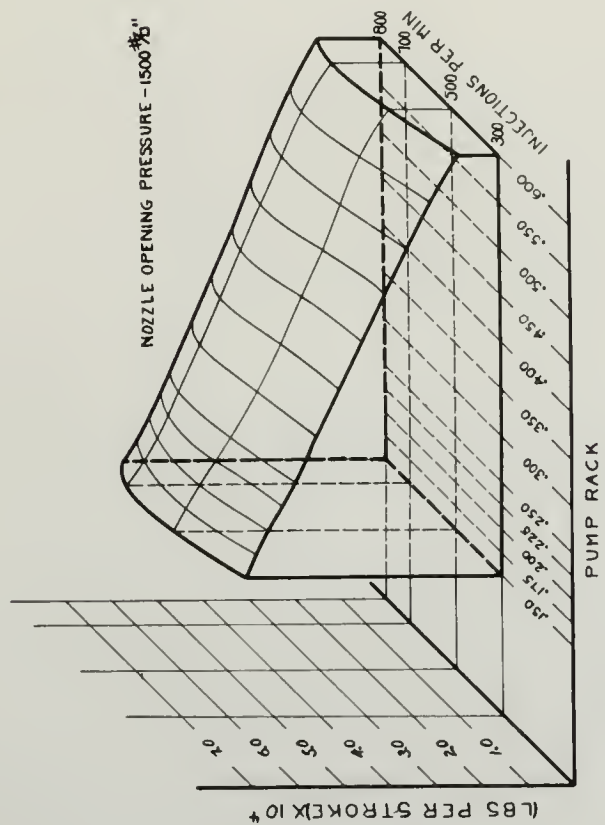
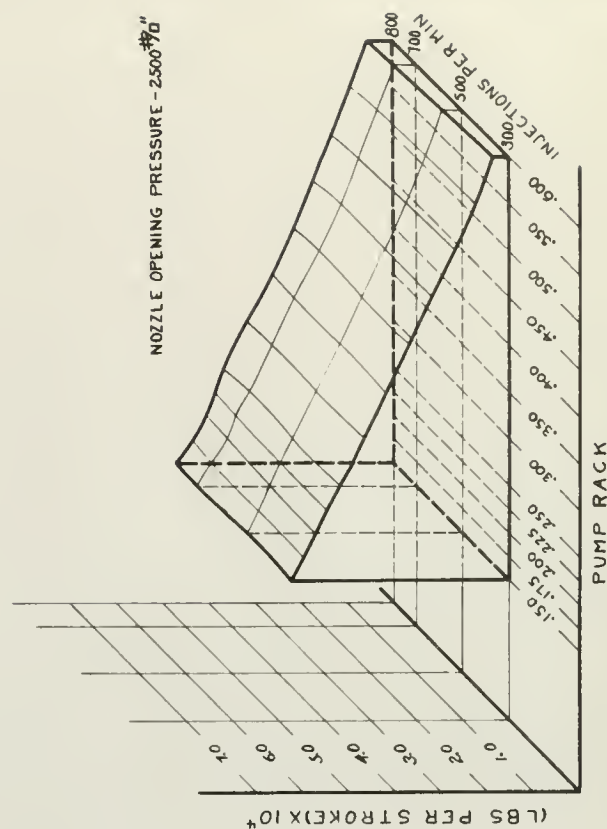
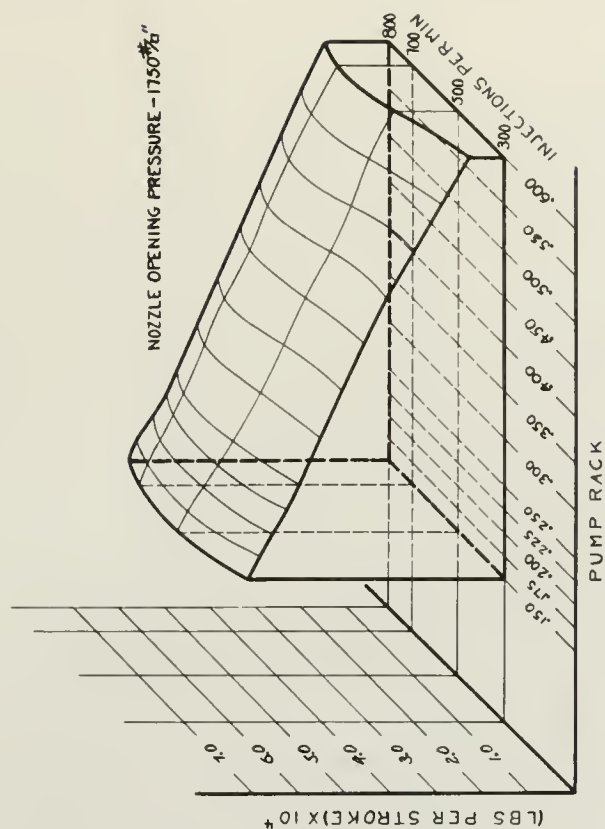


changes contour with both speed and pressure variations. Disregarding the dynamics of the system this surface should be coincident with the plane surface ABCD which represents the amount of oil which the pump should deliver based on displacement data. Obviously the planes A B C D in Figure 6 are horizontal as shown since the theoretical weight delivered per stroke is independent of all variables save rack setting. That the weight per stroke contour surface does not coincide with the plane surfaces A B C D Figure 6, is clearly shown for the rack settings considered (.200, .300, .400, .500). The fact that the pump actually delivers more (or less) oil than is shown by displacement computations may be explained in the following way. (1) (2) (6)*

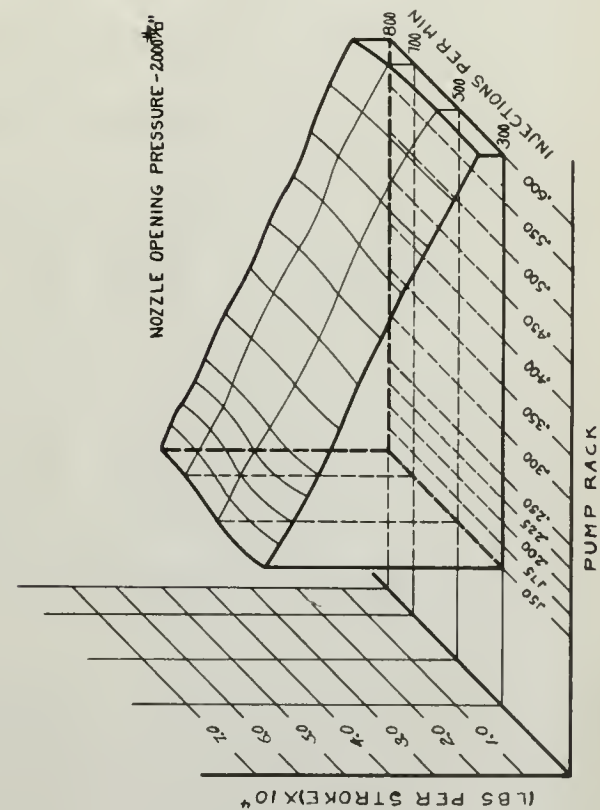
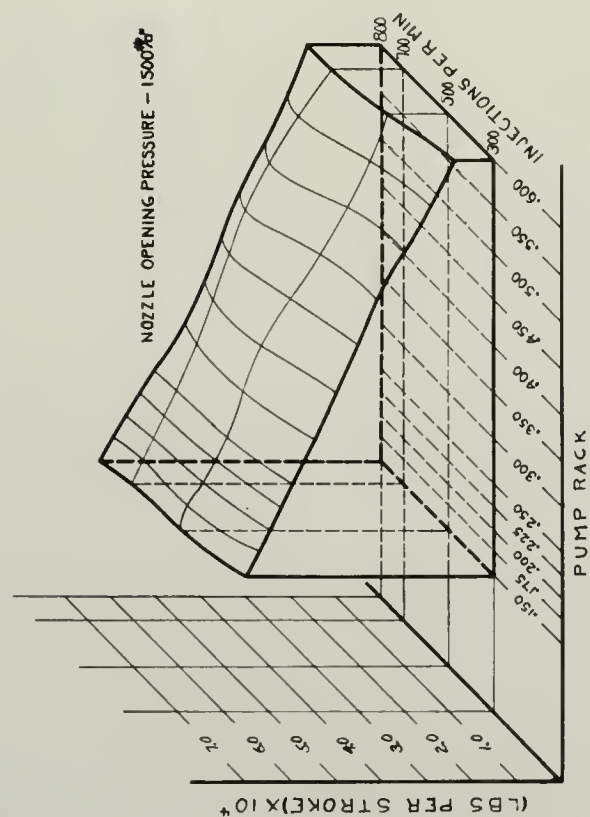
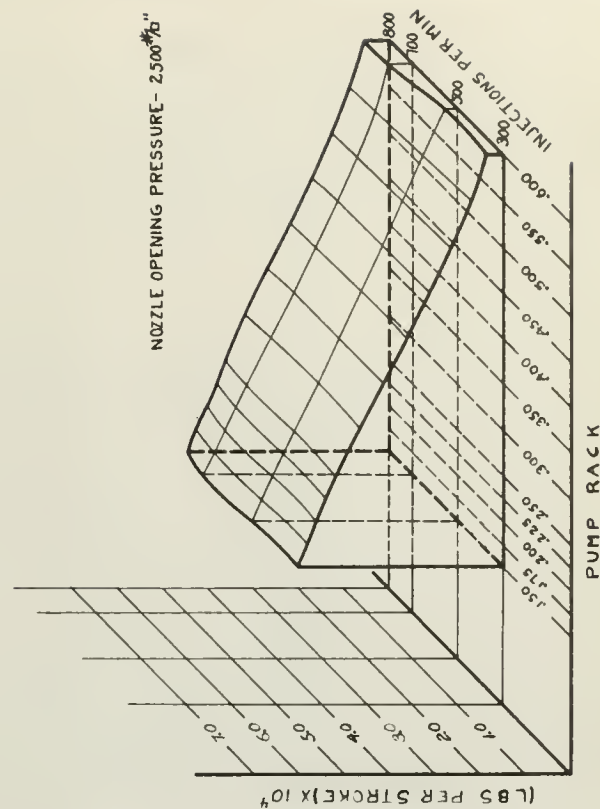
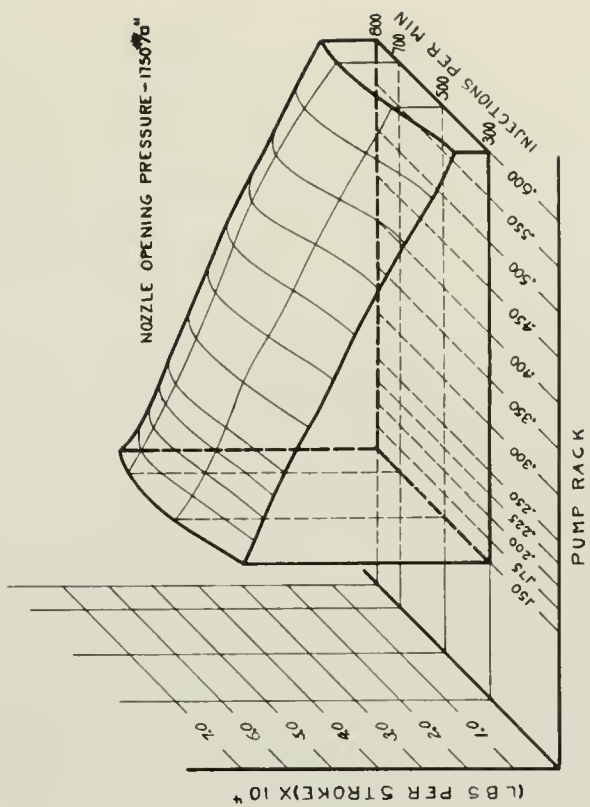
Once the check valve has been lifted off its seat, due to the oil pressure created by the plunger motion, oil will flow into the line and pressure waves will develop between the face of the pump plunger and the nozzle valve. When the pump reaches the point of release, the check valve will remain off its seat for an appreciable interval due to its inertia as well as to the friction force occasioned by the viscous drag of the oil flowing past the valve. Thus the next pressure wave, after release, reflected from the plunger face will force additional oil past the open check valve and into the fuel line. This action is possible, in spite of the fact that the plunger by-pass channel puts the pump chamber in direct communication with the suction side of the pump, since the by-pass channel area is so small that the pressure wave moving toward the plunger face will be reflected from the plunger face before it has driven any oil out of the chamber space into the suction line. After the check valve has been seated, pressure

*Such designations refer to similar numbers in bibliography.

THE INFLUENCE OF PUMP SPEED AND RACK SETTING
ON DISCHARGE RATE
FOR VARIOUS NOZZLE OPENING PRESSURES
PIPE LENGTH - 15 INCHES
FIGURE 7



THE INFLUENCE OF PUMP SPEED AND RACK SETTING
ON DISCHARGE RATE
FOR VARIOUS NOZZLE OPENING PRESSURES
PIPE LENGTH - 30 INCHES
FIGURE 8



surfaces A B C D. The amount of oil actually delivered decreased with increasing nozzle opening pressures due to the greater force exerted by the oil to keep the check valve on its seat against the action of the pressure surges, and to the increased pump leakage occasioned by the higher discharge pressures.⁽⁵⁾

Figure 7 shows the influence of rack setting and speed on the weight of oil discharged per stroke for various nozzle opening pressures and a pipe length of 15 inches; Figure 8 shows a series of similar surfaces for a pipe length of 30 inches. Again it may be noted from these figures that as the nozzle opening pressure approaches 2175 lbs. per sq. in., the weight contour surface approaches plane surfaces; see Figures 7 and 8 for nozzle opening pressures of 2000 and 2500 lbs. per sq. in. These contour surfaces also clearly show the influence of pump speed changes on the quantity of oil discharged. Here it is to be noted that in general the weight of oil discharged increased with higher speed up to about 700 R.P.M.⁽³⁾ while for the range between 700 -- 800 R.P.M., the weight of oil decreased;⁽⁵⁾ this may be seen most clearly in Figures 7 and 8 and Table II for nozzle opening pressures of 1500 and 1750 lbs. per sq. in. For the nozzle opening pressures of 2000 and 2500 lbs. per sq. in. (in the neighborhood of 2175 lbs. per sq. in.) the influence of pump speed was not so marked.

Table II follows.

recovered by the above described process, (8)

Figure 7 shows the influence of test weight and speed on the weight of oil discharged per stroke for various nozzle opening pressures and a pipe length of 16 inches; Figure 8 shows a series of similar curves for a pipe length of 32 inches. Again it may be noted from these figures that as the nozzle opening pressure approaches 1500 lbs. per sq. in., the weight number varies approximately as Figures 7 and 8 for nozzle opening pressures of 1000 and 1500 lbs. per sq. in. These curves indicate also clearly how the influence of pump speed changes as the opening of all discharges. Here it is to be noted that in general the weight of oil discharged increased with higher speed up to about 700 R.P.M. (4) also for the range between 700 -- 1000 R.P.M., the weight of oil discharged; (5) this may be seen more clearly in Figures 7 and 8 and Table II for nozzle opening pressures of 1000 and 1500 lbs. per sq. in. for the nozzle opening pressures of 1000 and 1500 lbs. per sq. in. (in the neighborhood of 1500 lbs. per sq. in.) the influence of pump speed was not so marked.

TABLE II

Rack Setting	Pump Speed	ACTUAL DISCHARGE (LBS. PER STROKE X 10 ⁰); Pipe Length 30".			
		Nozzle Opening	Pressure (Lbs. Per	Sq. In.)	
		1500	1750	2000	2500
.200	300	498	505	467	418
	500	541	535	485	412
	700	556	550	450	424
	800	568	508	457	403
.300	300	394	404	366	318
	500	459	452	365	311
	700	460	457	359	312
	800	435	422	361	319
.400	300	303	304	269	216
	500	352	337	265	216
	700	388	373	257	217
	800	355	336	270	217
.500	300	202	191	166	122
	500	226	236	173	123
	700	300	279	167	128
	800	254	228	174	132

The fact that the quantity of oil increased for increased pump speed up to 700 R.P.M., and then decreased for further speed increases may be explained as follows: higher pump speeds give higher plunger speed, and hence impart greater velocities to the oil being discharged from the pump; this increase in kinetic energy causes a greater quantity of oil to flow past the pump check valve after release and before the check valve has become seated. As the pump speed increased beyond a certain value, however, the volumetric efficiency of the pump decreased since oil could not flow into the pump chamber fast enough to completely fill it, and the quantity of oil discharged decreased.

A comparison of Figures 7 and 8 shows pipe length, for the two



Figure 9

SPRAY DEVELOPMENT FOR AVERAGE OF SEVERAL HUNDRED INJECTIONS.

Time Between Frames, .000278 seconds

Nozzle Opening Pressure, 1750 lbs. per sq. in.

Pump Speed, 600 R.P.M.

Length of Pipe Line, 15 inches

Rack Setting, .400 (2.40 x 10⁴ lbs. of oil per stroke)

Chamber Pressure, Atmospheric



Figure 10

SPRAY DEVELOPMENT FOR AVERAGE OF SEVERAL HUNDRED INJECTIONS.

Time Between Frames, .000278 seconds
 Nozzle Opening Pressure, 1750 lbs. per sq. in.
 Pump Speed, 600 R.P.M.
 Length of Pipe Line, 15 inches
 Rack Setting, .600 (.381 x 10⁴ lbs. of oil per stroke)
 Chamber Pressure, Atmospheric.

lengths investigated, to have little influence.⁽³⁾ For nozzle opening pressures of 1750 and 2500 lbs. per sq. in., the two pipe lengths gave almost identical contour surfaces, while for nozzle opening pressures of 1500 and 2000 lbs. per sq. in. some irregularities between the contour surfaces may be noted. These irregularities, as may be seen, occurring for pump speeds of 500 R.P.M. and above.

A comparison of Figures 9 and 10 show the influence of rack setting on penetration at a pump speed of 600 R.P.M., and a nozzle opening pressure of 1750 lbs. per sq. in., when discharging against atmospheric pressure. Figure 9 is for a rack setting of .400, while Figure 10 is for a rack setting of .600. For both rack settings, the pictures marked 1 were taken at two degrees of pump shaft angular displacement after the point of injection was observed which is equivalent to one eighteen-hundredth of a second. Thereafter the pictures were taken in succession at one degree pump shaft intervals (one thirty-six-hundredth of a second). In Figure 9, evidence of secondary discharges may be seen in pictures 4, 5, 13, 14, 15 and 17.⁽⁴⁾ Maximum penetration of 16.5 inches is shown in picture 12 and cut off in picture 13. The depth of penetration is quite uniform up to the point of cut off, picture 13. It is interesting to note at this point that Figure 9 shows a definite injection period over an interval of 13 degrees angular pump shaft displacement, whereas the table on page 12 for a rack setting of .400 shows the injection period to be only 10.25 degrees. This point again established the fact that the pump actually discharges more oil under certain conditions than is theoretically possible.

The first of these is the fact that the
 pressure of 1700 and 1800 lbs. per sq. in. is
 not a constant value, but varies with the
 position of the piston. This is due to the
 fact that the pressure is not uniform
 throughout the cylinder, but is higher
 at the bottom than at the top. This is
 due to the weight of the gas, which
 is not negligible. The pressure at the
 bottom of the cylinder is therefore
 higher than at the top. This is the
 reason why the pressure is not uniform
 throughout the cylinder.

The second of these is the fact that the
 pressure of 1700 and 1800 lbs. per sq. in.
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The third of these is the fact that the
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The fourth of these is the fact that the
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 is not a constant value, but varies with
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 higher than at the top. This is the
 reason why the pressure is not uniform
 throughout the cylinder.

The fifth of these is the fact that the
 pressure of 1700 and 1800 lbs. per sq. in.
 is not a constant value, but varies with
 the position of the piston. This is due
 to the fact that the pressure is not
 uniform throughout the cylinder, but is
 higher at the bottom than at the top.
 This is due to the weight of the gas,
 which is not negligible. The pressure
 at the bottom of the cylinder is therefore
 higher than at the top. This is the
 reason why the pressure is not uniform
 throughout the cylinder.

In Figure 10 it may be readily seen that the spray penetration is quite irregular. Secondary discharges may be noted in pictures 3, 4, 5, 6, 7, 9 and 10. Cut off has taken place in picture 7, and here the maximum penetration was about 10 inches. Again it may be noted in Figure 10 that injection took place over a pump shaft interval of 8 degrees whereas the table on page 12, for a rack setting of .600, shows this interval to be only 3 degrees.

It is not to be used for the purpose of determining

is quite possible. We must therefore be

2. 3. 4. 5. 6. 7. 8. 9. 10. 11. 12. 13. 14. 15. 16. 17. 18. 19. 20. 21. 22. 23. 24. 25. 26. 27. 28. 29. 30. 31. 32. 33. 34. 35. 36. 37. 38. 39. 40. 41. 42. 43. 44. 45. 46. 47. 48. 49. 50. 51. 52. 53. 54. 55. 56. 57. 58. 59. 60. 61. 62. 63. 64. 65. 66. 67. 68. 69. 70. 71. 72. 73. 74. 75. 76. 77. 78. 79. 80. 81. 82. 83. 84. 85. 86. 87. 88. 89. 90. 91. 92. 93. 94. 95. 96. 97. 98. 99. 100. 101. 102. 103. 104. 105. 106. 107. 108. 109. 110. 111. 112. 113. 114. 115. 116. 117. 118. 119. 120. 121. 122. 123. 124. 125. 126. 127. 128. 129. 130. 131. 132. 133. 134. 135. 136. 137. 138. 139. 140. 141. 142. 143. 144. 145. 146. 147. 148. 149. 150. 151. 152. 153. 154. 155. 156. 157. 158. 159. 160. 161. 162. 163. 164. 165. 166. 167. 168. 169. 170. 171. 172. 173. 174. 175. 176. 177. 178. 179. 180. 181. 182. 183. 184. 185. 186. 187. 188. 189. 190. 191. 192. 193. 194. 195. 196. 197. 198. 199. 200. 201. 202. 203. 204. 205. 206. 207. 208. 209. 210. 211. 212. 213. 214. 215. 216. 217. 218. 219. 220. 221. 222. 223. 224. 225. 226. 227. 228. 229. 230. 231. 232. 233. 234. 235. 236. 237. 238. 239. 240. 241. 242. 243. 244. 245. 246. 247. 248. 249. 250. 251. 252. 253. 254. 255. 256. 257. 258. 259. 260. 261. 262. 263. 264. 265. 266. 267. 268. 269. 270. 271. 272. 273. 274. 275. 276. 277. 278. 279. 280. 281. 282. 283. 284. 285. 286. 287. 288. 289. 290. 291. 292. 293. 294. 295. 296. 297. 298. 299. 300. 301. 302. 303. 304. 305. 306. 307. 308. 309. 310. 311. 312. 313. 314. 315. 316. 317. 318. 319. 320. 321. 322. 323. 324. 325. 326. 327. 328. 329. 330. 331. 332. 333. 334. 335. 336. 337. 338. 339. 340. 341. 342. 343. 344. 345. 346. 347. 348. 349. 350. 351. 352. 353. 354. 355. 356. 357. 358. 359. 360. 361. 362. 363. 364. 365. 366. 367. 368. 369. 370. 371. 372. 373. 374. 375. 376. 377. 378. 379. 380. 381. 382. 383. 384. 385. 386. 387. 388. 389. 390. 391. 392. 393. 394. 395. 396. 397. 398. 399. 400. 401. 402. 403. 404. 405. 406. 407. 408. 409. 410. 411. 412. 413. 414. 415. 416. 417. 418. 419. 420. 421. 422. 423. 424. 425. 426. 427. 428. 429. 430. 431. 432. 433. 434. 435. 436. 437. 438. 439. 440. 441. 442. 443. 444. 445. 446. 447. 448. 449. 450. 451. 452. 453. 454. 455. 456. 457. 458. 459. 460. 461. 462. 463. 464. 465. 466. 467. 468. 469. 470. 471. 472. 473. 474. 475. 476. 477. 478. 479. 480. 481. 482. 483. 484. 485. 486. 487. 488. 489. 490. 491. 492. 493. 494. 495. 496. 497. 498. 499. 500. 501. 502. 503. 504. 505. 506. 507. 508. 509. 510. 511. 512. 513. 514. 515. 516. 517. 518. 519. 520. 521. 522. 523. 524. 525. 526. 527. 528. 529. 530. 531. 532. 533. 534. 535. 536. 537. 538. 539. 540. 541. 542. 543. 544. 545. 546. 547. 548. 549. 550. 551. 552. 553. 554. 555. 556. 557. 558. 559. 560. 561. 562. 563. 564. 565. 566. 567. 568. 569. 570. 571. 572. 573. 574. 575. 576. 577. 578. 579. 580. 581. 582. 583. 584. 585. 586. 587. 588. 589. 590. 591. 592. 593. 594. 595. 596. 597. 598. 599. 600. 601. 602. 603. 604. 605. 606. 607. 608. 609. 610. 611. 612. 613. 614. 615. 616. 617. 618. 619. 620. 621. 622. 623. 624. 625. 626. 627. 628. 629. 630. 631. 632. 633. 634. 635. 636. 637. 638. 639. 640. 641. 642. 643. 644. 645. 646. 647. 648. 649. 650. 651. 652. 653. 654. 655. 656. 657. 658. 659. 660. 661. 662. 663. 664. 665. 666. 667. 668. 669. 670. 671. 672. 673. 674. 675. 676. 677. 678. 679. 680. 681. 682. 683. 684. 685. 686. 687. 688. 689. 690. 691. 692. 693. 694. 695. 696. 697. 698. 699. 700. 701. 702. 703. 704. 705. 706. 707. 708. 709. 710. 711. 712. 713. 714. 715. 716. 717. 718. 719. 720. 721. 722. 723. 724. 725. 726. 727. 728. 729. 730. 731. 732. 733. 734. 735. 736. 737. 738. 739. 740. 741. 742. 743. 744. 745. 746. 747. 748. 749. 750. 751. 752. 753. 754. 755. 756. 757. 758. 759. 760. 761. 762. 763. 764. 765. 766. 767. 768. 769. 770. 771. 772. 773. 774. 775. 776. 777. 778. 779. 780. 781. 782. 783. 784. 785. 786. 787. 788. 789. 790. 791. 792. 793. 794. 795. 796. 797. 798. 799. 800. 801. 802. 803. 804. 805. 806. 807. 808. 809. 810. 811. 812. 813. 814. 815. 816. 817. 818. 819. 820. 821. 822. 823. 824. 825. 826. 827. 828. 829. 830. 831. 832. 833. 834. 835. 836. 837. 838. 839. 840. 841.

Since the maximum likelihood was shown to converge, only 10 iterations were required to obtain the maximum likelihood estimates. Table 11 lists the

There is a large amount of information available on the Internet, and it is important to be able to find it. The following are some of the most useful resources for finding information on the Internet:

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...and the

CONCLUSIONS

1. From Figure 6 it may be concluded that for this system there exists a certain nozzle opening pressure for which a linear relation exists between weight of fuel discharged per stroke and rack setting, and further that this linear relation is independent of speed for the range investigated; that is to say, for a pump speed from 300 -- 800 R.P.M. and a nozzle opening pressure of from 1500 -- 2500 pounds per sq. in. In this case the nozzle opening pressure required to give this linear relation was found to be about 2175 pounds per sq. in.

It appears reasonable to suppose that a similar nozzle opening pressure will exist, and may be found, for any injection system of the type investigated.

2. A comparison of Figures 7 and 8 shows that for the pipe lengths investigated there was little or no change in the weight of oil delivered per stroke for the two pipe lengths.

3. Figures 6, 7 and 8 show definitely that the quantity of oil discharged per stroke decreases with increase in the nozzle opening pressure.

4. Figures 6, 7 and 8 show that when operating at a nozzle opening pressure other than 2175 pounds per sq. in., as mentioned in 1 above, the weight of oil discharged per stroke will vary with pump speed, and the greater the departure of the pressure

[illegible]

It appears reasonable to suppose that a stable matrix
would preserve all order, and may be used, for any purposes
connected with the page investigation.

1. A comparison of the two sets of data for the year 1960 and 1961 shows that the number of cases of disease has increased in the United States.

all diagnosed for stress disorder with interest in the study

It is not possible to determine the exact date of the first meeting of the committee, but it is known that the committee was organized in the summer of 1917.

from this value of nozzle opening pressure, the greater will become the influence of pump speed. Generally speaking, the quantity of oil delivered per stroke, for the system investigated, increased with higher pump speed up to about 700 R.P.M., and further speed increase caused the quantity of oil discharged to decrease.

5. The quantity of oil delivered per stroke follows in a general way the rack setting, as seen from Figures 7 and 8, but this relation is not linear unless the nozzle is set to open at the proper spring setting (2175 pounds per sq. in. for this system); see Figure 6.

6. For a moderate load (rack setting of .400), it may be noted from Figure 9 that the spray penetration is reasonably uniform and that there is little evidence of secondary discharges. However, for light loads (rack setting of .600), it may be seen from Figure 10 that the spray penetration is irregular and that there is much evidence of secondary discharges.

from this series of results appears to be that the quantity of the influence of each factor, especially frequency, the quantity of all factors for which, for the purpose of statistical treatment, this factor may be taken up as a unit, the factor which has been used the quantity of all factors for which.

2. The quantity of all factors for which follows in a general way the same pattern, as seen from figures 1 and 2, but this relation is not linear, which has been seen in the proper order of the factors (the factors are not in the same order as in figure 1).

3. For a moderate load (load rating of 100), it may be said from figure 3 that the frequency relation is reasonably linear and that there is little evidence of secondary changes. However, for light loads (load rating of 100), it may be seen from figure 3 that the frequency relation is irregular and that there is some evidence of secondary changes.

4. The results of the present study are in general agreement with the results of other studies, but there are some differences. The results of the present study are in general agreement with the results of other studies, but there are some differences. The results of the present study are in general agreement with the results of other studies, but there are some differences.

ACKNOWLEDGMENT

The author desires especially to thank Professor C. J. Vogt of the Department of Mechanical Engineering, University of California, who started this investigation and who has given untiringly and cheerfully of both his time and wisdom in order that this problem might be carried forward.

Thanks are also extended to Lieutenant H. S. Persons, U.S.N., who assisted in obtaining all the necessary data and the taking and developing of the pictures contained herein; and to Mr. J. E. Gullberg, Lecturer in the Department of Zoology, University of California, for the assistance given in the taking of many pictures.

Finally, I desire to thank members of the W. P. A. Project in the Department of Mechanical Engineering for their work in constructing the experimental station and making necessary alterations.

APPENDIX

The author has been especially fortunate in the cooperation of the Department of Industrial Engineering, University of California, Berkeley, who have provided him with the necessary facilities and equipment for the study of the problem and observation of the life and work of the worker in the factory along his entire career.

Thanks are also due to the University of California, Berkeley, for the assistance in obtaining all the necessary data and for the facilities of the physical sciences building and to the University of California, Berkeley, for the assistance in the study of the problem. The author is also indebted to the University of California, Berkeley, for the assistance in the study of the problem.

Finally, I desire to thank the University of California, Berkeley, for the assistance in the study of the problem and for the facilities of the physical sciences building and to the University of California, Berkeley, for the assistance in the study of the problem.

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APPENDIX

- (1) The following is a list of the names of the persons who have been appointed to the various committees of the House of Representatives, since the year 1800.
- (2) The following is a list of the names of the persons who have been appointed to the various committees of the Senate, since the year 1800.
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